



Performance of a prototype Stirling domestic refrigerator

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ARTICLE INFO

Article history:

Received 22 June 2007

Accepted 16 February 2008

Available online 18 March 2008

Keywords:

Stirling cycle

Simulation

Performance

V-type integral Stirling refrigerator

ABSTRACT

The V-type integral Stirling refrigerator (VISR) consisting of an expansion cylinder, a compression cylinder and a heat exchanger in-between was developed and tested in this research. The expansion and compression pistons are V-shaped and driven by a crank-shift mechanism. A cold head connected to the expansion piston provides the cooling capacity. The refrigerator has a semi-hermetic configuration. The parameters such as the power consumption and the coefficient of performance (COP) are investigated under various rotating speeds and charged pressures. The results would help us for the optimal design and operation of the V-type integral Stirling refrigerator, when it is applied to the domestic refrigeration system. The characteristics curves of the prototype VISR could be applied to other usages.

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1. Introduction

In the past decades the Stirling cycle cryocoolers have been developed rapidly because of its high efficiency, fast cool-down, small size, light weight, low power consumption, high reliability, and so on. And now the Stirling cycle cryocooler is widely used as a cooler for infrared sensors and high temperature superconducting devices and also applied to the domestic and commercial refrigerators due to its environmental friendliness and applicability to the wide temperature range.

Domestic refrigeration system used Stirling cycle was first studied by Finkelstein and Polonski [1], and the optimum performance parameters for the simple Stirling cycle refrigerator and heat pumps have also been presented as a function of operating conditions [2–7]. A free-piston Stirling cooler (FPSC) with Rankine compressor has been investigated [8–10]. And a portable cooler box was discussed [11].

The Stirling cycle is fundamentally different from the Rankine cycle used in conventional refrigerators. No phase change occurs in Stirling cycle. The food is kept below $-30\text{ }^{\circ}\text{C}$ for long time and the conventional vapor-compression machines cannot cool-down up to $-20\text{ }^{\circ}\text{C}$. The cost for installation of the conventional one is rising because of necessity of the double-stage compressor for attaining $-30\text{ }^{\circ}\text{C}$ [12].

A V-type integral Stirling refrigerator (VISR) was developed for pursuing low cost and easy manufacture. The VISR is different from the split-type free-displacer Stirling refrigerator (STFSR) regarding the driving mechanism of the displacer. The displacer of the VISR is directly driven by a motor through a crank-shaft mechanism. In

case of the STFSR, the displacer is driven by the fluid force or gaseous pressure which is indirectly induced by the piston motion and the stroke is thus variable and depends upon the operating conditions.

So far, some analysis had been done on the V-type Stirling refrigerator. Ataer et al. [13] analyzed the V-type Stirling cycle refrigerator using the control volume method. Huang and Chen [14] derived a transfer-function for the integral-type Stirling refrigerator (ITSR). Walker and co-worker [15,16] simulated the performance of the Stirling cryocoolers with two-phase, two-component working fluid. They applied the V-type Stirling cycle refrigerator to the cryogenic temperature range. The V-type Stirling cycle applicable to the domestic refrigeration systems are also investigated about the effect of the working fluid on its performance but only helium not nitrogen. The purpose of this work is to investigate the thermal performance of the V-type Stirling cycle refrigerator (VISR) used helium or nitrogen as a working fluid. The difference between us and previous researches is that our temperature range is over -20 to $-60\text{ }^{\circ}\text{C}$, and the effect of nitrogen is also investigated.

In this prototype, the cold head is installed between the expansion port and the regenerator and the electric heater covers the cold head. The compression and expansion pistons are driven by a crank-shaft mechanism functioning with a fixed phase between compression and expansion processes. Using helium or nitrogen as a working fluid, the thermodynamic process was optimized by the theoretical analysis and computer simulation. And the simulation results were compared with experimental ones.

2. Description of the prototype refrigerator

It is well known that the working fluid of the Stirling cycle is gas and no phase-transition occurs during the cycle. The ideal Stirling

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cycle consists of two isothermal and two isochoric heat transfer processes where the isochoric heat transfer takes place in a regenerator. The gas absorbs thermal energy from the environment during the isothermal expansion and releases heat to the environment during the isothermal compression process. Therefore, the performance of a Stirling cycle cooler would be determined by the temperatures at the warm and cold heads.

The VISR consists of the expansion room, the cold head, the regenerator, the cooler, the compression room, the expansion and compression pistons, and a slider-crank mechanism which is shown in Figs. 1 and 2.

The design specifications of the prototype Stirling refrigerator are listed in Table 1 and the diameter and length of the regenerator are described in Table 2. The charged pressures are varied to inves-

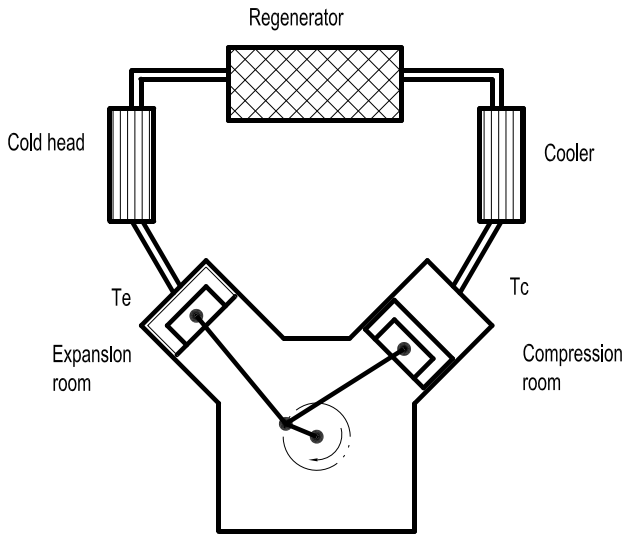


Fig. 1. Schematics of the Stirling cycle refrigerator.



Fig. 2. The photo of the Stirling cycle unit.

Table 1

Design specifications of the V-type Stirling cycle refrigerator

Compression piston diameter	65 mm
Compression piston stroke	50 mm
Expansion piston diameter	40 mm
Expansion piston stroke	50 mm
Expansion piston length	110 mm
Wire screen material of regenerator	Stainless steel
Wire screen mesh no. of regenerator	200
Wire diameter of regenerator	0.1 mm
Cold-end temperature	238 K

Table 2

The specifications for three configurations of the regenerator

Configurations	1	2	3
Regenerator length (mm)	80	60	50
Regenerator diameter (mm)	55	50	40

tigate the effects of the configuration difference on the performance.

3. Development of the model

3.1. Description of basic model

The isothermal model of the Stirling refrigerator developed by Schmidt is appropriate to add the effects of the additional components, because the Schmidt model is well known. The model would be subsequently improved for taking account of a pressure drop, shuttle heat loss, etc. They are calculated separately and inserted into the results described by Smith to fit the analyzed result to the data [17].

The system design analysis of the VISR can be carried out assuming the followings:

1. The gas temperatures in the compression room and gas passage are uniform and thus the regenerator inlet temperature T_{ri} is assumed to be equal approximately to the gas temperature in the compression room T_c and in the passage T_t , i.e. $T_{ri} \approx T_t \approx T_c$.
2. The gas temperature T_e in the expansion room is approximately equal to the outside wall temperature T_L of the cold head, i.e. $T_e \approx T_L$.
3. The wall temperatures at both sides of the regenerator, T_{ri} and T_{ro} , are approximately constant. Thus, $T_w \approx T_{ri} \approx T_t \approx T_c$; $T_{ro} \approx T_e \approx T_L$.
4. The working fluid behaves as an ideal gas, $PV = RT$.
5. The regenerator is an energy-storage element made from packed wire screens. The flow is 1D flow and there is no axial heat transfer.

Taking into pressure drop through the system due to flow resistance, the average pressure of starting operation p_{av} is supposed to be equal to the charged pressure p_{ch} . But pressures at the respective components are depending on the pressure drop. The nomenclatures are used along the reference [17]. The expansion volume V_e is assumed to be changed sinusoidally.

$$V_e(\phi) = \frac{V_E}{2} \cdot [1 + \cos(\phi)] \quad (1)$$

where V_E is the swept volume of the expansion room and ϕ is the crank angle,

$$\phi \equiv \frac{2 \cdot \pi \cdot t}{\tau} \quad (2)$$

The volume in the compression room V_c is given as follows:

$$V_c(\phi) = \frac{w \cdot V_E}{2} \cdot [1 + \cos(\phi - \alpha)] \quad (3)$$

where w is the ratio of the swept volume in the compression room to that in the expansion room and α is the phase angle between V_e and V_c . In addition, there is some dead volume V_d in the regenerator. S is the dimensionless dead volume, τ the temperature ratio in the compression and expansion room, θ minimal pressure angle, δ parameter, which are given as follows

$$S = \frac{T_c}{V_E} \sum \frac{V_{dj}}{T_{dj}}, \quad (4)$$

$$\tau = \frac{T_c}{T_E}, \quad (5)$$

$$\theta = \tan^{-1} \left[\frac{w \cdot \sin(\alpha)}{\tau + w \cdot \cos(\alpha)} \right], \quad (6)$$

$$\delta = \frac{\sqrt{\tau^2 + 2 \cdot \tau \cdot w \cdot \cos(\alpha) + w^2}}{\tau + w + 2 \cdot S}. \quad (7)$$

The pressure ratio is expressed in the following:

$$p_{\max}/p_{\min} = (1 + \delta)/(1 - \delta) \quad (8)$$

The average pressure in the system p_{av} is presented

$$p_{av} = p_{\min} \cdot \sqrt{\frac{1 + \delta}{1 - \delta}} \quad (9)$$

Therefore, the instantaneous pressure in the system $p(\phi)$ is also given by the average pressure p_{av}

$$p(\phi) = p_{av} \cdot \frac{\sqrt{1 - \delta^2}}{1 + \delta \cdot \cos(\phi - \theta)} \quad (10)$$

3.2. Development of the model

The total pressure drop across the system is given by the summation of the frictional flow resistance and bends as follows [18]:

$$\Delta p = \sum \Delta p_m + \Delta p_l \quad (11)$$

The flow resistance of the respective components is given as follows:

$$\Delta p_m = \xi \frac{\bar{v}_p^2}{2} \rho \quad (12)$$

where ρ is a gas density, ξ a flow resistance coefficient, $\bar{v}_p$ a mean gas flow velocity in the components.

At the respective outlets of the compression room, the regenerator, the cold head, the cooler, and the expansion room, the gas is compressed. And thus ξ is given as follows:

$$\xi = 0.5 \left(1 - \frac{A_2}{A_1} \right)^2$$

At the respective inlets of the expansion room, the regenerator, the cold head, the cooler, and the expansion room, the gas is expanded. So ξ is given as follows:

$$\xi = \left(1 - \frac{A_2}{A_1} \right)^2$$

where A_1 describes a sectional area of the compression room, the regenerator, the cold head, the cooler, and the expansion room, and A_2 expresses a sectional area of each outlet and inlet pipes.

The flow resistance of pipe (Δp_l) is given as follows:

$$\Delta p_l = \lambda \frac{\bar{v}_p^2}{2} \rho \frac{l}{d_i} \quad (13)$$

where λ are a frictional coefficient, l a pipe length, and d_i its inner diameter. The λ is given as follows:

$$\lambda = \frac{1}{\left(1.14 + 2 \lg \frac{d_i}{\Delta} \right)^2}$$

where Δ is the roughness value of a pipe wall, and $\Delta = 0.1$ – 0.5 , depending on materials.

And cooling capacity Q_E was calculated by following formula:

$$Q_E = f \int_0^{2\pi} p_e(\phi) dV_e = \frac{n}{60} p_{eav} V_E \frac{\delta \sin \theta}{1 + \sqrt{1 - \delta^2}} \quad (14)$$

$$Q_C = -\tau Q_E \quad (15)$$

$$W = -Q_E - Q_C = (\tau - 1) Q_E \quad (16)$$

where f is the rotational frequency of the crank, and n the rotational speed.

The net cooling capacity Q_{net} of the VISR can be evaluated by subtracting the heat losses from the available cooling capacity. The heat losses are given by the heat conduction loss Q_{cond} of the regenerator, the regenerator's enthalpy flow loss Q_{enth} , the shuttle heat loss Q_{shutt} of the expansion piston and the system flow resistance loss Q_{res} .

$$Q_{\text{net}} = Q_E - Q_{\text{enth}} - Q_{\text{cond}} - Q_{\text{shutt}} - Q_{\text{res}} \quad (17)$$

The losses Q_{enth} , Q_{cond} and Q_{shutt} can be calculated according to the reference [12]. The loss of Q_{res} is calculated by the following formula:

$$Q_{\text{res}} = \Delta p \frac{V_E n}{30} \quad (18)$$

where n is the rotational speed.

The actual input power N_{ac} is described as follows:

$$N_{ac} = \frac{W}{\eta_T \eta_m \eta_{el}} \quad (19)$$

where η_T is the isothermal efficiency of the machine, η_m the mechanical efficiency of the machine and η_{el} the motor efficiency of the machine.

The COP of the VISR can be evaluated by the following equation:

$$\text{COP} = \frac{Q_{\text{net}}}{N_{ac}} \quad (20)$$

4. Experimental research

4.1. Test system

The prototype VISR was constructed for investigating its performance and power consumption. The temperatures at the compression and expansion rooms, at the inlet- and outlet-ports of regenerator and their inter-space were measured with the calibrated PT100 thermocouples. The pressures were monitored with Piezo-electric dynamic pressure sensors installed at the outlet-ports of the compression and expansion rooms. The discharged pressure at the crankcase was measured by the pressure gauge with 0.001 MPa accuracy which was calibrated by a dead weight tester. The rotational speed and shaft power were measured with a torque sensor. The input voltage, current and the power consumptions of the cooler were recorded with accuracies of $\pm 0.1\%$, $\pm 0.2\%$ and $\pm 1\%$, respectively. The wattmeter and uninterrupted power supply were used and the frequency was changed by the transducer. The real time data were recorded by the National Instruments M Series data acquisition (DAQ) system, i.e. NI-DAQ 8.1. In order to minimize the heat flow from the ambient to the cold head, the heater was insulated with polyurethane foam. Fig. 3 shows the schematic diagram of the experimental system.

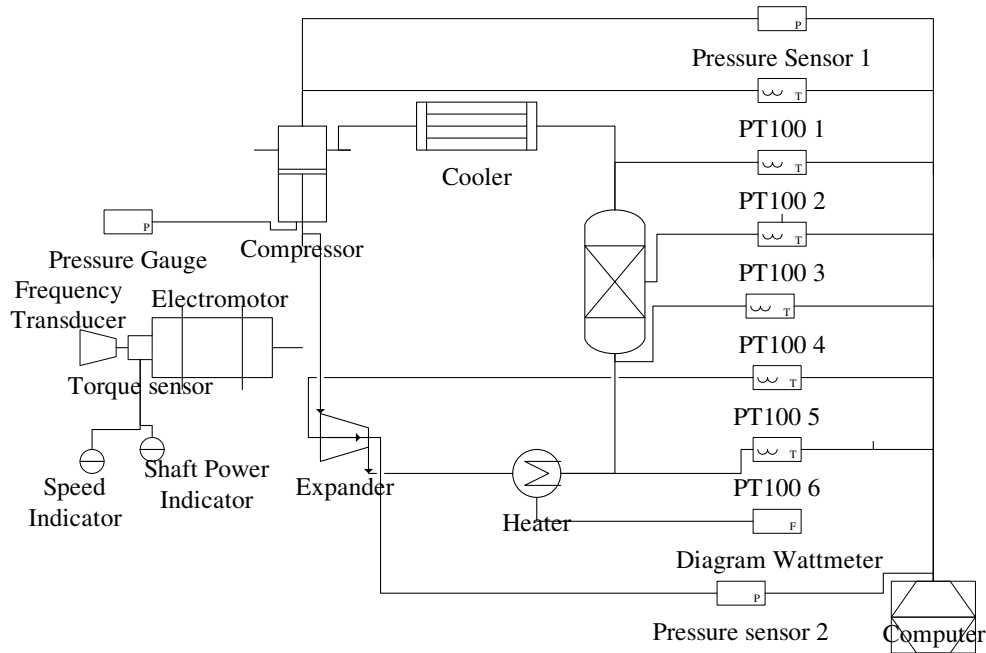


Fig. 3. Schematic diagram for experiments of the VISR.

4.2. Experiment procedure

Using helium and nitrogen as working fluids, the rotational frequency and charged pressure are changed in the experiments under the three kinds of configurations given in Table 2. The charged pressures of helium and nitrogen in the refrigerator were changed from 0.25 MPa to 1.4 MPa and the rotational frequency was 10–40 Hz.

5. Results and discussions

Fig. 4 shows the variation of the compression and expansion pressures with the crank angle when helium or nitrogen is charged at 1.0 MPa. And the rotational speed is 600 rpm in case of Conf. 1 (shown in Table 2).

In Fig. 4, the pressure difference between the compression and the expansion means the pressure loss. And the pressure loss for nitrogen is larger than that of the helium.

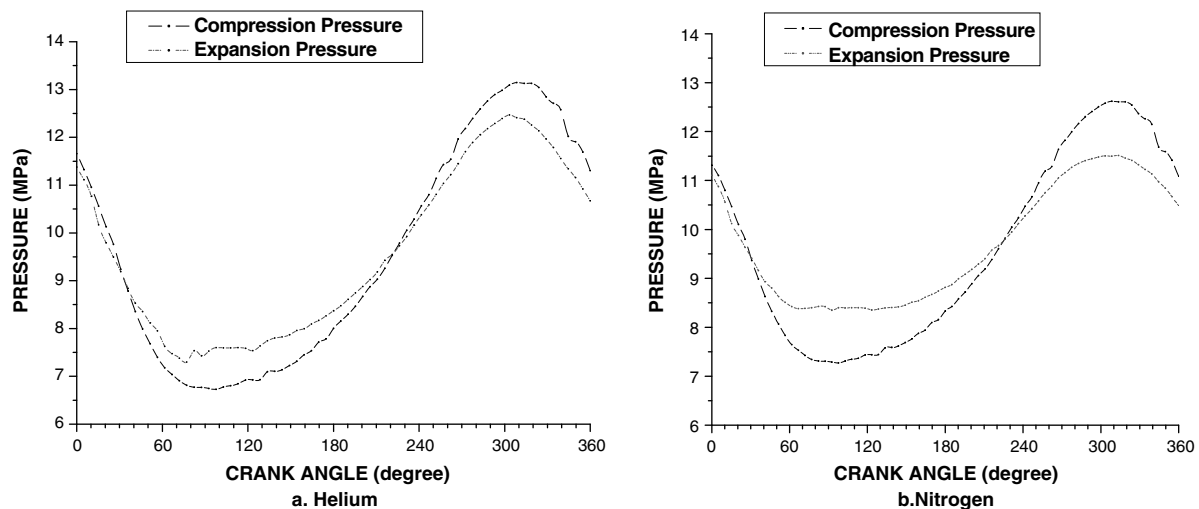


Fig. 4. Pressure variation in the compression and expansion rooms with the crank angle: (a) Helium and (b) Nitrogen.

Fig. 5 shows the simulation and experiment results of the cooling capacity and COP as a function of the rotational speed under the Conf. 1 in which the pressure of nitrogen or helium is changed at 1.0 MPa. It is observed that in case of helium the cooling capacity increases with rotational speed but the COP has a maximum. In case of nitrogen, both of the cooling capacity and the COP have an optimum. And in cases of nitrogen, the optimum rotational speeds for the cooling capacity and the COP are in the range of 500–700 rpm. But for helium, the optimum rotational speeds for COP are at 800–1000 rpm. The cooling capacity of helium is greater than that of nitrogen, because the specific heat capacity of helium is larger than that of nitrogen, and the flow resistance of helium is much smaller than nitrogen. The difference in optimum points between nitrogen and helium is caused by the difference of the total pressure drop across the system from the compression room to the expansion room and its total pressure drop of nitrogen is larger than that of helium. This pressure drop is in proportion to the square of working fluid velocity and a mass flow. The mass flow

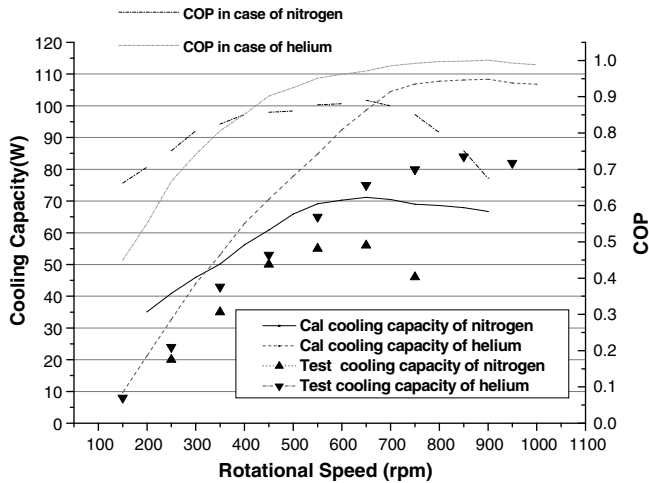


Fig. 5. Variations of cooling capacity and COP with rotational speed.

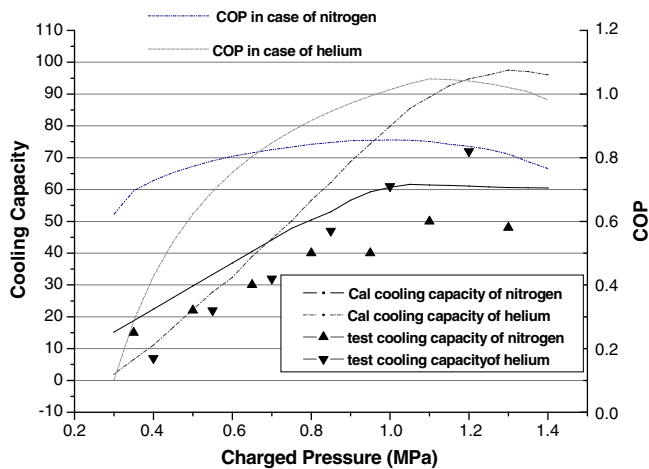


Fig. 6. Variations of cooling capacity and COP with charged pressure, and Cal in the figure caption means calculated and Conf. the configuration in the Table 2.

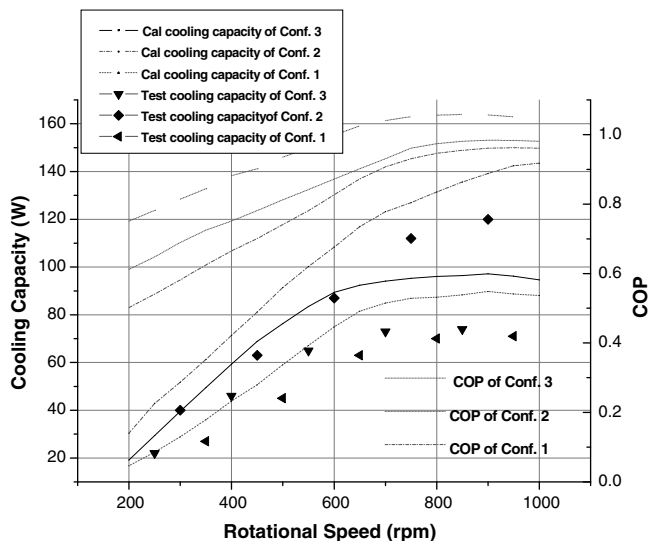


Fig. 7. Cooling capacity and COP as function of rotational speed.

of nitrogen is larger than that of helium. We investigated the influence of the charged pressure of both helium and nitrogen on the performance, i.e. at 0.4, 0.5, 0.6, 0.7, 0.9 MPa. The trend of cooling capacity and COP against charged pressures was similar to the case of 1.0 MPa, although their relative values are different at the different charged pressure.

Fig. 6 shows the dependence of the cooling capacity on the charged pressure under the Conf. 1 at 800 rpm rotational speeds. It is shown that the cooling capacity increases with the charged pressure up to 1.0 MPa in case of nitrogen and up to 1.3 MPa in case of helium. Fig. 7 shows the cooling capacities in case of 1.0 MPa helium, under the Conf. 1, Conf. 2 and Conf. 3. It is observed from Fig. 7, that the Conf. 2 is the optimal configuration even in simulation.

We can see from Figs. 6 and 7 that the experimental cooling capacity is about 80% of that of the calculation. The manufacturing defects in the machine would cause the deterioration of the performance compared with the predicted ones. And the inner leakage of the working fluid from the working space to the crankcase causes also the decrease of cooling capacity.

6. Conclusions

Based on the simulation and experiments the following could be concluded:

1. To obtain the optimal configuration for each VISR components in the refrigeration system, each component must be matched each other to satisfy the required specifications.
2. It is shown that the cooling capacity increases with the charged pressure up to 1.0 MPa in case of nitrogen and up to 1.3 MPa in case of helium. The Conf. 2 is shown to be the optimal configuration for the cooling capacity. The result shows that helium is better than nitrogen.
3. The optimum rotational speed for the cooling capacity is different between nitrogen and helium. The COP has a peak value around 600 rpm for nitrogen, and for helium around 900 rpm.
4. These results show that the VISR is applicable to the domestic refrigeration.

Acknowledgements

The authors are grateful to Prof. He Yaling, Prof. Takeshi KAWAI and Assistant Prof. Liu Yingwen of Xi'an Jiaotong University for their valuable comments.

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